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EFFECT OF STATOR-ROTOR AXIAL CLEARANCE ON COLD-AIR PERFORMANCE OF A TURBINE WITH TRANSPIRATION-COOLED STATOR BLADING

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ABSTRACT

A single-stage axial-flow turbine having transpiration coolant ejection through wire-mesh-shell stator blades was tested with the stator-rotor axial clearance increased to 1 1/2 inches (3.81-cm) from its design value of 1/2 inch (1.27-cm). Results of the two turbine configurations are compared over a range of coolant fractions from 0 to 7 percent, and for turbine conditions corresponding to equivalent design speed and work. It was found that turbine performance improved with the larger clearance. This suggests that if transpiration-cooled stator blades are to be used, adequate stator-rotor clearance be provided.

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SUMMARY

A cold-air experimental investigation was conducted on a 30.0-inch (0.762-m) single-stage research turbine to determine the changes in turbine aerodynamic performance of coolant flow through wire-mesh shell stator blades when the rotor was moved 1-inch (2.54-cm) further downstream than design. These effects were determined over a range of coolant fraction from 0 to 7 percent for turbine operation at design equivalent speed and a specific work output (based on primary air) of 17.00 Btu per pound (39 572 J/kg). Results, in terms of efficiency changes, for this modified turbine are compared to those previously obtained for the same turbine with the design stator-rotor axial clearance (1/2 in.; 1.27 cm).

It was found that moving the rotor downstream resulted in improved turbine efficiency over the range of coolant fraction investigated. The stage efficiency increased an almost constant one percent (from 90 to 91 percent). A 2-percent improvement in rotor component efficiency was also noted. The latter is believed attributable to the fact that the additional mixing of the thick blade wakes from the transpiration-type stator blades tested resulted in improved rotor entry conditions of these low velocity fluids. And this resulted in the gain in stage efficiency. It was concluded that if similar transpiration-cooled stator blades are to be used, with appreciable amounts of low velocity fluid in the wake regions, consideration should be given to provide ample stator-rotor blade axial clearance.

INTRODUCTION

Advanced engines for aircraft propulsion are utilizing higher turbine inlet temperatures to meet their performance requirements. These temperatures in some cases necessitate cooling the turbine blades. The general method of cooling turbine blading is to bleed air from the compressor, direct this air through the turbine blading, and then discharge it into the main gas stream.

The NASA Lewis Laboratory has conducted a series of tests on a single-stage, 30-inch tip diameter research turbine to determine the effect of stator-blade coolant ejection. First, a turbine having plain (uncooled) stator blades was tested (refs. 1 to 3) in order to establish a performance level to use as a basis of comparison. This stator was then replaced with three different air-cooled blading configurations, all with the same blade profile shape. The first cooled stator consisted of hollow blades with a slot provided in the trailing edges for cooling-air discharge. The other two simulated transpiration cooled blades, one having a self-supporting shell with discrete holes, and the other having a wire-mesh skin supported by an inner strut structure. A complete description of the blades tested as stator cascades as well as with the rotor are presented in references 4 to 10. The results of the program are briefly summarized in reference 11.

The reference stator studies indicated that the transpiration-type stators had significantly larger losses than that found with trailing-edge coolant discharge. Also, stage tests at zero coolant flow showed that the performance of turbines with porous stator blades had deteriorated more than would be theoretically expected from these increased stator losses. Calculations of rotor efficiencies were therefore made to determine the effect of circumferential variations in flow conditions from the stators on rotor performance. Reference 11 indicated a significant difference in rotor efficiency between the various stator blades tested. It was felt that if more axial spacing were provided between the stator blades and the rotor blades, more complete mixing of the low-momentum stator coolant flow with the primary flow would prevail, resulting in better air-entry conditions to the rotor. Accordingly, the wire-mesh transpiration-cooled stator blades were installed in the test facility with the stator-rotor axial clearance increased from the design value of 1/2 inch (1.27 cm) to 1 1/2 inches (3.81 cm). This report presents the results of stage tests of this modified turbine.

Cold-air tests were conducted at the equivalent design speed and over a range of pressure ratio encompassing the equivalent specific design work output (17.00 Btu/lb; 39 572 J/kg). Coolant fraction (ratio of coolant flow to primary flow) was varied from 0 to 7 percent. Obtained performance results are compared to those obtained for the turbine having the same stator and the design axial stator-rotor spacing (1/2 in.; 1.27 cm).

The turbine tests were conducted with the primary air maintained at a constant nominal inlet stagnation pressure of 30 inches of mercury absolute (10.16 N/cm^2). Both the primary and coolant air were supplied by the laboratory combustion air system at a nominal temperature of 543° R (303 K). Thus, the heat-exchange effects that occur in cooled turbines are not simulated. Only the aerodynamic effects are considered herein.

APPARATUS AND INSTRUMENTATION

The turbine tested was a single-stage axial-flow cold-air research turbine, with a tip diameter of 30 inches (0.762 m) and a blade length of 4 inches (0.102 m). These blades are characterized by thick profiles, blunt leading and trailing edges, and low solidity, typifying the features required of a turbine for high-temperature application. The design procedure used to evolve the blade shapes was discussed in reference 1. The design requirements of the turbine are summarized as follows:

Equivalent specific work output, Btu/lb (J/kg)	17.00 (39 572)
Equivalent mean blade speed, ft/sec (m/sec).	500 (152.4)
Equivalent mass flow, lb/sec (kg/sec).	39.9 (18.10)

The subject investigation is an extension of the tests reported in references 9 and 10, wherein the design stator was replaced with blades having the same profile (within manufacturing tolerances), but made of stainless steel wire-mesh shell material supported by a central strut. The blade fabrication technique is described in the references. For completeness, a photograph depicting this technique is shown in figure 1. A closeup view of a blade is shown in figure 2, with an enlarged view of the wire mesh and a typical shell-strut electron-beam weldment. The stator assembly comprises 50 of these transpiration-cooled type blades. The assembly is shown in figure 3.

The stator was installed in the test facility along with the same rotor (fig. 4) that has been used throughout the stage-test program (refs. 3, 6, 8, and 10). For the subject investigation, however, the rotor was relocated 1-inch (2.54 cm) further downstream from the stator than heretofore. Mechanically, this was accomplished by providing a 1-inch (2.54-cm) spacer between the rotor and the shaft, and a corresponding 1-inch (2.54-cm) spacer at the inner wall behind the stator blade row. These modifications are shown on the facility cut-away sketch, figure 5. The insert of figure 5 shows the "as designed" stator-rotor axial clearance (1/2-in.; 1.27-cm).

The instrumentation was the same as that described in reference 10, in which the stage performance of the turbine with wire-mesh stator blading and design stator-rotor axial clearance was determined. Briefly, total and static pressures and temperatures were measured at the turbine inlet. Static pressures were measured at the stator exit. At the turbine exit, total and static pressures were measured, along with the outlet flow angles. The measuring stations are shown in figure 5. In addition, turbine rotative speed was measured with an electronic counter in conjunction with a magnetic pickup and a sprocket secured to the rotor shaft. Turbine torque was measured using a strain-gage-type load cell. The load cell and the readout system were calibrated before and after each day's testing.

All instrumentation was connected to a 100-channel data acquisition system which measured and recorded the electrical signals from the appropriate transducers. At each data point, five readings of each transducer were recorded and subsequently numerically averaged.

PROCEDURE

The performance of the turbine with wire-mesh porous stator blades and design stator-rotor axial clearance (1/2-in.; 1.27-cm) is reported in reference 10. The second phase of these tests was conducted at the equivalent design speed and over a range of pressure ratio bracketing the equivalent specific design work output of 17.00 Btu per pound (39 572 J/kg). Stator blade coolant fraction was varied from 0 to 7 percent. This resulted in blade cavity pressures both below and above the turbine-inlet pressure.

As stated previously, the turbine was modified for the subject investigation by relocating the rotor 1 1/2 inches (3.81 cm) downstream of the rotor as contrasted to the design axial stator-rotor clearance of 1/2 inch (1.27 cm). Data were obtained from this modified turbine configuration in the same manner and over the same range of turbine variables as reported in reference 10, phase II. Both turbines were operated with the turbine-inlet primary-air total state conditions of 30.0 inches of mercury absolute (10.16 N/cm^2) and approximately 545^o R (303 K). Overall pressure ratios were varied by adjusting the turbine-exit pressure.

Turbine inlet total pressure was calculated from the static pressure and total temperature measured at station 1 (fig. 5), along with the measured primary mass flow and the known annulus flow area. The outlet total pressure was calculated by using static pressure, the combined primary and coolant mass flows, the annulus area, the area-averaged turbine outlet flow angle, and the total temperature at station 3 (fig. 5). Equations for these pressures are presented in the stage tests of references 3, 6, 8, and 10.

Comparative results for the turbines with wire-mesh stator blades and with two different stator-rotor blade axial clearances are presented in terms of stage primary efficiency, stage thermodynamic efficiency, and rotor efficiency. The primary efficiency relates the total power of both fluids (as measured by the dynamometer) to the ideal power of only the primary flow. The thermodynamic efficiency is a measure of the loss characteristics when considering the ideal energies of both the primary and coolant flows. The rotor efficiency expresses the specific work output of the total flow (primary + coolant) through the rotor, to the ideal work. A more thorough discussion of these efficiencies, and the methods of calculating them, is described in reference 10.

RESULTS AND DISCUSSION

Cold-air tests were made on a single-stage turbine equipped with wire-mesh transpiration-cooled stator blades and with the rotor relocated 1 1/2 inches (3.81 cm) downstream of the stator blades. Data were cross-plotted and are presented herein in terms of stage primary efficiency, stage thermodynamic efficiency, and rotor efficiency for conditions corresponding to equivalent design speed and a turbine primary specific-work output of 17.00 Btu per pound (39 572 J/kg). The latter corresponds to the equivalent design work of the base (plain-bladed) turbine. A range of coolant fraction from 0 to 7 percent was investigated. Results are compared with those previously obtained for the turbine with design (1/2-in.; 1.27-cm) stator-rotor axial clearance (ref. 10).

Stage primary efficiency. - A comparison of the stage primary efficiency for the two turbine configurations with different stator-rotor axial clearances is presented in figure 6. It is apparent that both efficiency curves closely parallel each other, are relatively insensitive to coolant flow changes, and the turbine with the increased stator-rotor axial clearance yielded an efficiency improvement of about one point. It was stated in reference 11 that, for the wire-mesh stator, when the coolant pressure inside the blade was equal to the turbine inlet pressure, the coolant fraction was 2 percent. Above this value of coolant flow rate, the stage primary efficiency (fig. 6) increased from about 90 to 91 percent as a result of moving the rotor further downstream. Obtained efficiencies, however, were lower than the 92.3 percent obtained for the base (uncooled) turbine (ref. 3), as indicated by the square on figure 6.

Stage thermodynamic efficiency. - The thermodynamic efficiencies for the turbines with the two different stator-rotor axial clearances are shown in figure 7. Again, both curves closely parallel each other, the efficiency decreasing in a parabolic fashion with increasing coolant flow. The modified turbine yielded the higher efficiency over the range of coolant flow tested. The difference in efficiency for the two turbine configurations, however, decreased with increasing coolant flow. For example, at low coolant flows, the efficiency for the modified turbine was about one point higher than the efficiency for the turbine with design (1/2 in.; 1.27 cm) stator-rotor axial clearance. At coolant flows above 3 percent, this differential efficiency remained relatively constant at about 1/2 point.

Rotor efficiency. - Rotor efficiencies for the two turbines with different stator-rotor axial clearances are shown in figure 8 as a function of coolant flow rate. The figure indicates that although rotor efficiency for both turbine configurations generally tended to decrease with added coolant, there was about a 2-point efficiency gain as a result of increasing the axial stator-rotor clearance from 1/2 inch (1.27 cm) to 1 1/2 inches (3.81 cm).

Stage primary and thermodynamic efficiencies are contingent upon stator efficiency as well as rotor efficiency. The efficiency of the wire-mesh stator was determined in the investigation of reference 9. That rotor efficiency was improved with the larger clearance, then, can be attributed to the fact that the combined coolant and primary flows leaving the stator mixed more thoroughly, resulting in less severe rotor entry losses. And this, in turn, was reflected in the observed improvement in stage performance (figs. 6 and 7).

CONCLUDING REMARKS

The preceding discussion has indicated an improvement in turbine performance was obtained when the rotor was moved 1 inch (2.54 cm) further downstream of the transpiration-type stator blades than design. This finding should not be construed to be a general phenomenon, but rather unique to this type of stator coolant ejection configuration. The results imply, however, that where mechanical considerations permit, the performance of turbines with this type of stator cooling scheme may be improved by allowing large axial blade row clearances. It is recognized that large axial clearances may increase the wall-friction losses, but these losses were apparently more than offset by the improvement resulting from better mixing of the coolant and primary flows before entering the rotor.

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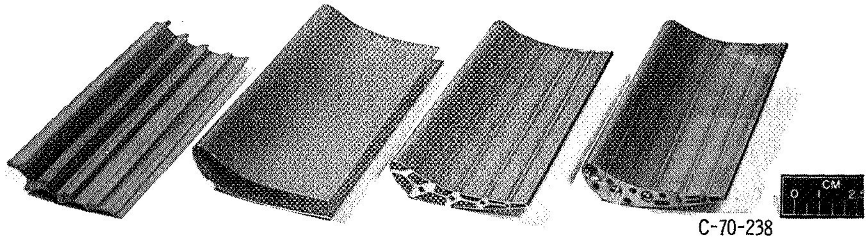

(a) Core. (b) Shell. (c) Welded blade. (d) Assembled blade.

Figure 1. - Wire-mesh shell blade parts and assembly.

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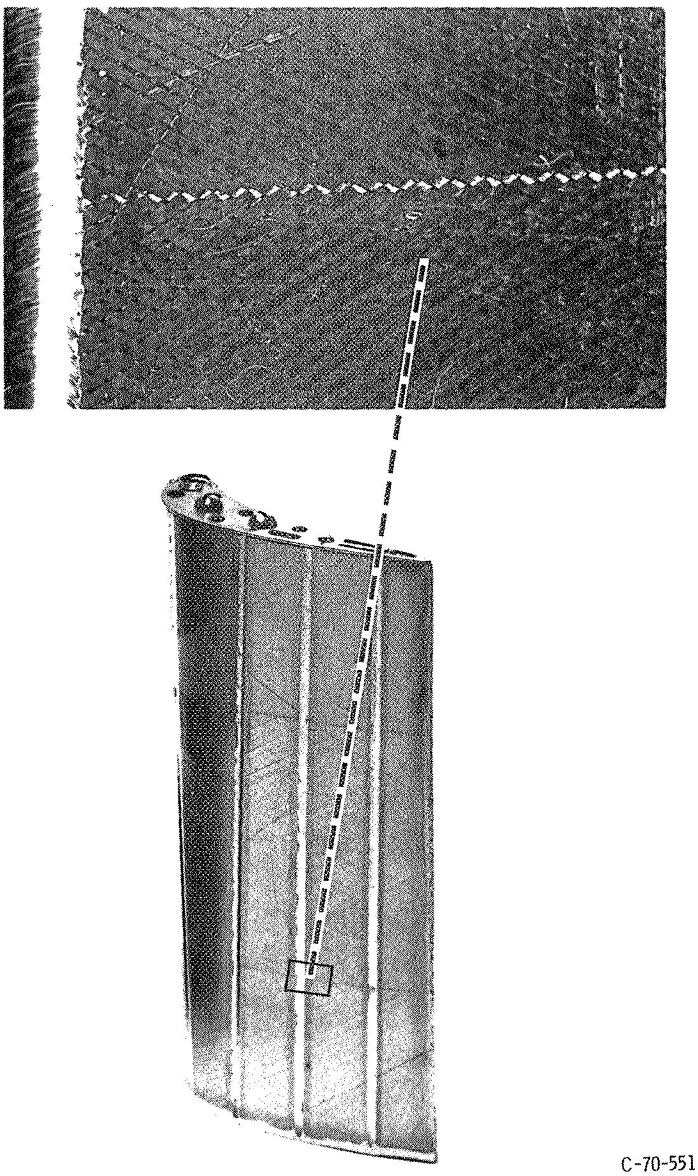

Figure 2. - Wire-mesh shell blade and magnification of shell material.

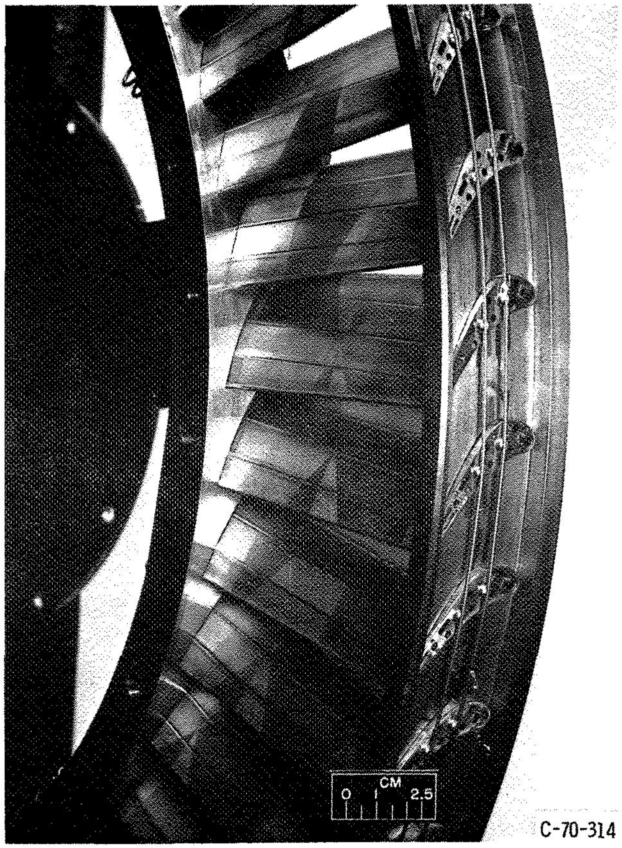

Figure 3. - Stator assembly of wire-mesh transpiration-cooled blades.

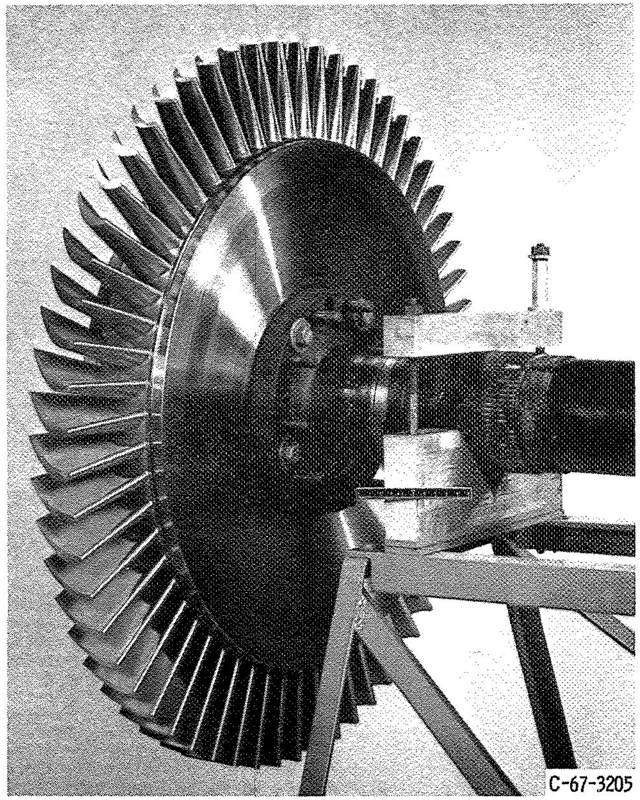

Figure 4. - Turbine rotor assembly.

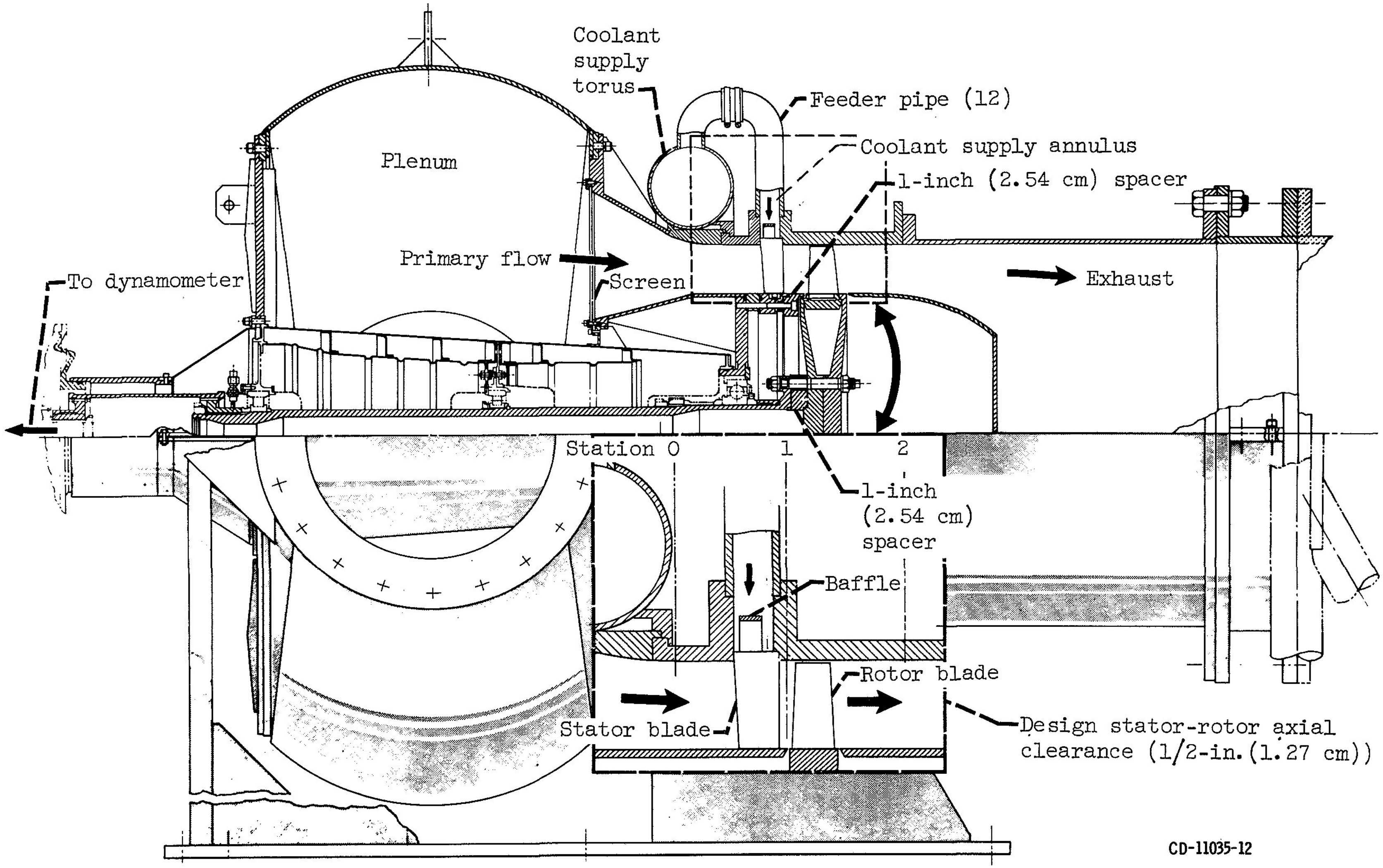

Figure 5. - Turbine test section.

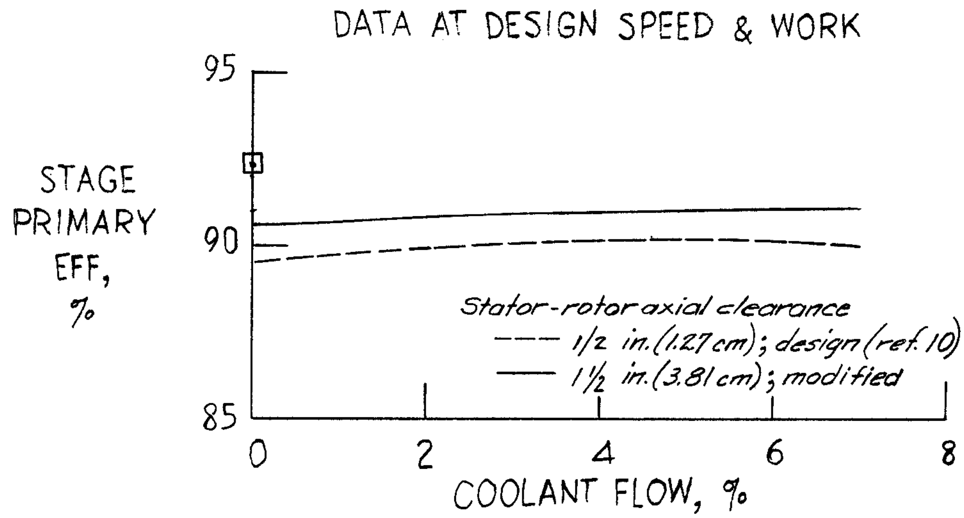

Figure 6.-Effect of stator-rotor axial clearance on turbine stage primary efficiency.

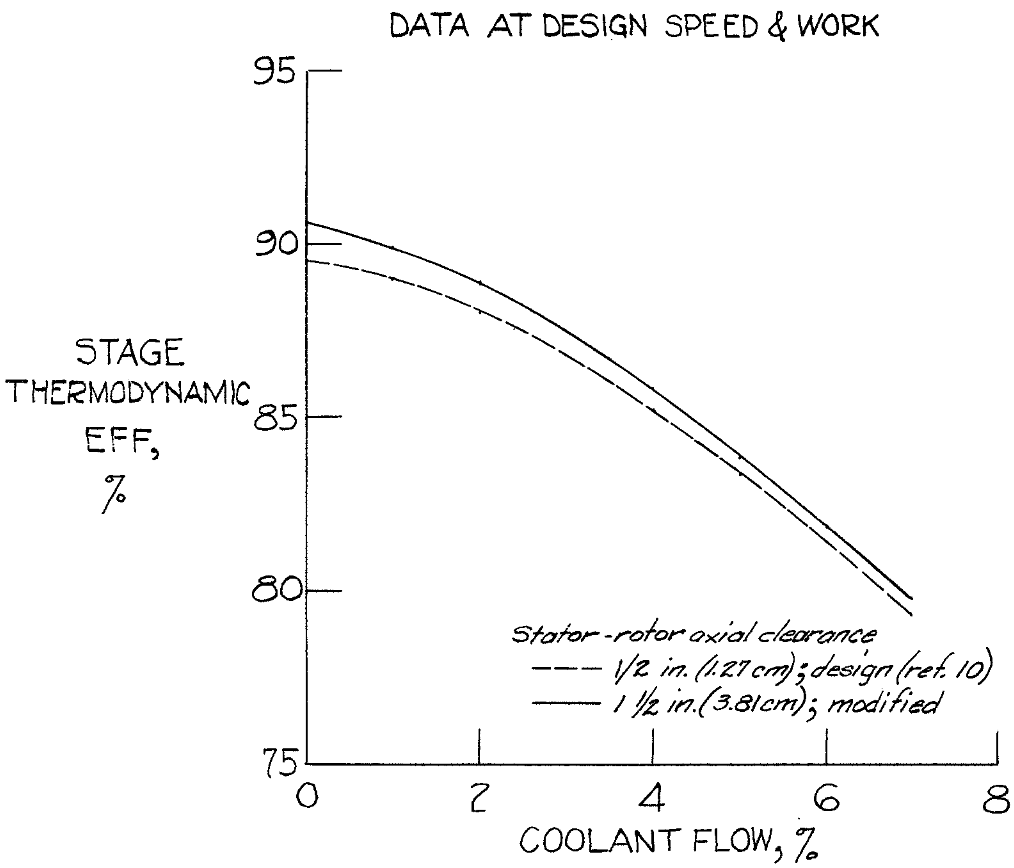

Figure 7.- Effect of stator-rotor axial clearance on turbine stage thermodynamic efficiency.

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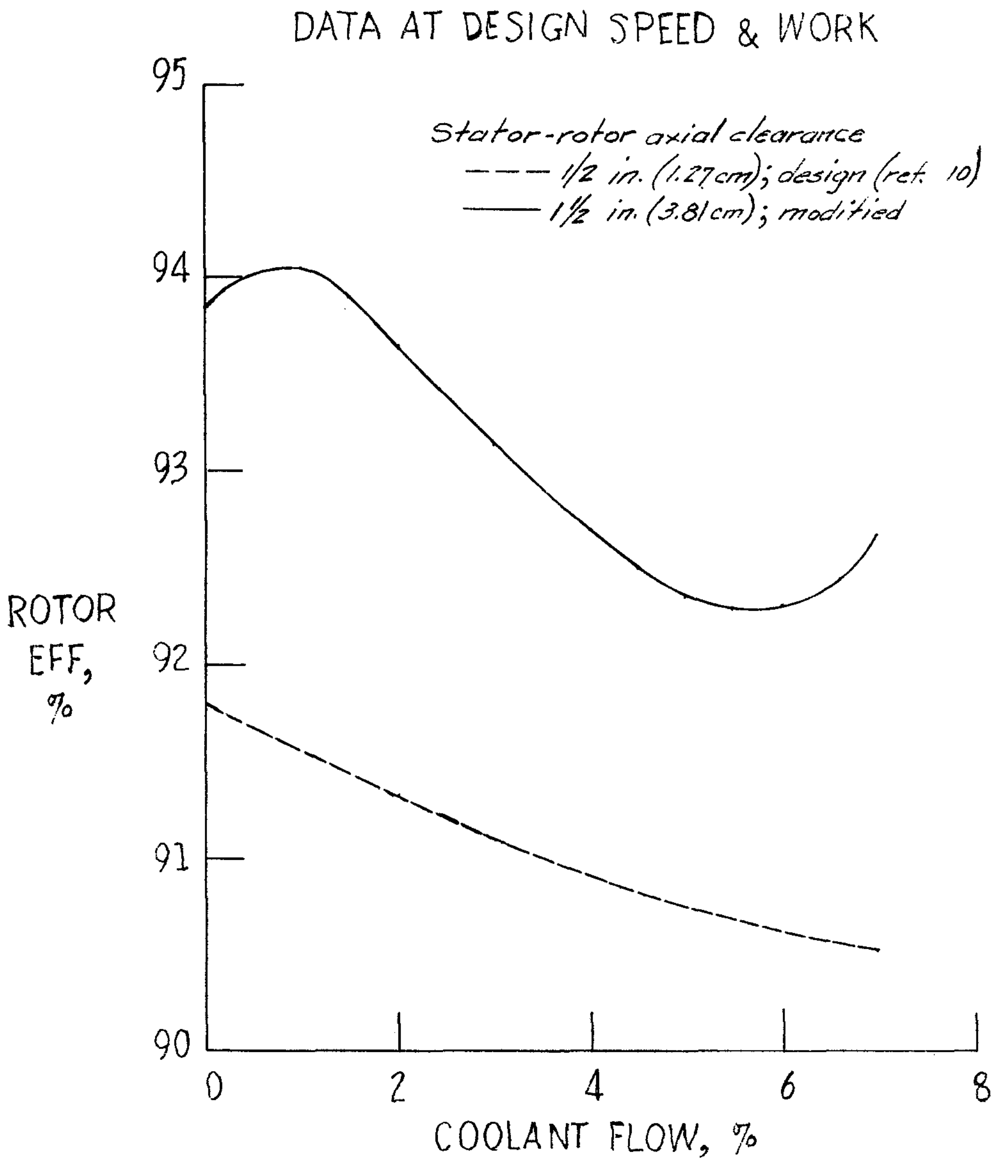

Figure 8. - Effect of stator-rotor axial clearance on rotor efficiency.